# Marine Labs*
# P&ID - theory and calculations †‡

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March 4, 2013

## 1 Introduction

**Abstract**

In any process plant or in marine systems we come across ***'Pipes'*** or ***'Tubes'*** quite often. These are the long hollow cylindrical geometric shapes used either for transfer of heat or fluid. ***'Pipes'*** are employed for transfer of fluids and ***'Tubes'*** are used in transfer of heat. Apart from the naming convention, these differ in technical specification as well[1]. When people started employing them into use, they followed certain nomenclature, terms, specifications particular to the region. Over time, with industrialization these developed to as national standards. Some attempt to unified standards world wide is also attempted and it is largely done by *ISO*.

The major piping standards which are commonly used in design of piping systems are:

1. **ASME** - American Society of Mechanical Engineers
2. **ISO** - International Standards Organisation
3. **DIN** - German Standards (similar to unified European standards)
4. **JIS** - Japanese Industrial Standards

*LaTeX typeset on GNU Emacs-Auctex
†Thanks to the Linux, LaTeX & open source community
‡**Spread knowledge in 'Open Source'**
[1] You can find a lot of explanation on this subject in my notes - Introduction to Piping, Standards & Terms

5. **EN** - Euro Norme (based on DIN standards)

Some of the other standards often referred are: **API**-*American Petroleum Institute*, **AWS**-*American Welding Society*, **MSS**-*Manufacturers Standardization Society*, **NACE**-*National Association of Corrosion Engineers*, **SAE**-*Society of Automotive Engineers*, **ASTM**-*American Society of Testing & Materials* together with national standards of individual countries like **IS**-*Indian Standards*, **ANSI**-*American National Standards Institute*

## 1.1 Looking into codes

Codes for piping standards gives us a frame work to which our design and specification must meet. Broadly codes cover subjects like:

1. ***Scope of the code*** - It contains a brief guidance on the code, introduction to any terms contained, sometimes legality of application of the code.

2. ***Design conditions*** - The subject may contains information related to design parameters. Usual parameters like Design pressures, temperature, factors affecting their choice, consideration of loads, allowable stresses & list of materials. The section also specifies allowable allowances & limits.

3. ***Design Pressure*** - The subject may contains information on methods of calculation to meet the required conditions. An elaborate steps of calculations are provided to arrive at the design pressure for the piping section to which this code is being applied. Also the piping section fittings,components & joinings are specified.

4. ***Stress & Flexibility*** - The subject provides information to make sure that the stresses in the piping section do not exceed limits. The loads conisidered in the section are external loads other than the pressure inside the piping section. External loads like -

   (a) Thermal Expansion of the piping section

   (b) Gravity loads on the section due to its own weight

   (c) Effects of vibrations, wind, earthquakes imparted to the piping section

   (d) Piping supports

5. ***Selection of Materials*** - As the section name specifies, this subject covers the choice of materials for fabrication of the piping section. Sometimes list of materials prohibited for use along with factors like high temperature or low temperature materials, corrosion, abrasion factors are discussed.

6. ***Applicable standards*** - Sometimes a code may consider another standard as equivalent in applicability. So such equivalent standards may be listed in this subject.

7. ***Fabrication & Fitting*** - This section of the code describes the fabrication methods and fitting up the sections in place. Choice of joining of sections along with limitations to the use some methods like threaded couplers are discussed.

8. ***Inspection,Examination & Testing*** - These requirements ensure that the finished piping section meets the design conditions. The methods offer a check for fabricator, Owners for their own assessment of the quality of the finished section.This offers a confirmation of the integrity of the piping section. Also this section specifies the responsibility of fabricator and owner in the area of testing, inspection & examination. The manufacturer,fabricator or a contractor is responsible for examination & documentation. Owner is responsible for inspection of the work.

## 2 Sections of the ASME code of interest !

- **B31.1**This code covers the essential requirements for construction of power and auxiliary piping systems for electric generation, industrial plants & marine applications. The code also applies to boilers, high temperature & high pressure piping systems.

- ASME B31.2[2] *(ANSI/NFPA Z223.1)* This code covers the fuel gas piping systems,appliances,equipment and related accessories.

- **B31.3** This code covers the process piping - Chemical & Petroleum process plants including refineries and related industries.The fluids handled by these plants are also covered in the code like compressed air, gas, steam, refrigerants etc.

- **B31.4** This code covers Tranportation pipeline sytems of Liquid hydrocarbons.

- **B31.5** This code covers Refrigeration & Heat transfer piping systems.

- **B31.8** This code covers Gas Transmission and Distribuition Piping systems.

[2]No more in use, it is now ANSI/NFPA Z223.1

## 3 Piping System

**Piping system** consists of pipes[3], fittings[4], valves for transportation of fluids.

**P&I Diagram** shows the arrangement of piping system together with the equipments to which they serve. It includes instrumentation provided, filters,strainers, counters, meters, pumps, valves, steam traps, sight gass, spools or other isolation arrangements & process equipment. All the components are drawn in approximate scale with every individual entity begin identified by a scheme. The parts of the diagram are drawn using standardised symbols, which facilitates easy and correct interpretation across globe. Overall a P & ID shows the process in operation.

There are some set of standard symbols for representing all the parts of process piping system. Refer German standards DIN 28004, DIN(1988), which have a wide acceptance in the industry.

## 4 Nature of Fluids !

**Rigid Body** A body in which the constituent particles always retain the same position with respect to one another. It will have a definite shape & size. In nature rigid bodies do not exist, but for our experiments with physics we assume some rigid bodies. We can approximate some bodies in real as rigid bodies.

**Elastic Body** means a body in which the constituent particles can change position with respect to each other. Such bodies can change their size or shape.

**Perfect Fluid** A perfect fluid is a substance whose shape can be altered by application of any small tangential force, applied long enough. Between different layers of the perfect fluid there is no rubbing or frictional force. Such fluids are non existant.

> *If a thin lamina is pushed edgeways through water the resistance to its motion along its surface is very small. There are no fluids in which this force vanishes. A hypothetical fluid in which this resistance force is assumed zero is a perfect fluid.*

[3] Refer my notes on pipes - *Introduction to Piping, Standards & Terms*

[4] The bends, miters & such fittings which change the direction of layout of the pipes

**Fluids** Fluids are subdivided into **Liquids** and **Gases** depending on whether the fluid is compressible(*or expandable*) or incompressible[5] on application of pressure alone, keeping temperature constant.

**Liquids** are substances like water whose total volume cannot be increased or diminished by application of force alone (*pressure*) however large it may be, but even a small force can change its shape. Another distinctive property of liquids over gases is that they form a free surface under gravitational fields & gravity has significant influence on liquids.

**Gases** are substances like air which can be easily made to change their volume on application of force. Gases do not form a free surface under gravitational field, generally gravitational effects are negligible for gases.

**Viscuos Fluids** *These are real fluids* Real fluids offer resistance to forces which tend to alter their shape. The layers of the fluids in contact offers resistance to shear. If this resistance to shear cannot be neglected, it is a *viscous fluid.*

**Pressure at a point** is the compressive stress at the point in a static fluid.In fluids the pressure(*compressive stress*) acts perpendicularly to any element within. In liquids pressure at a depth below the free surface consists of equal pressures in all directions *(Pascal's law)*. The pressure due to liquid alone above this point at a depth 'h' is

$$P_{(due\ to\ liquid\ depth)} = \rho g h \tag{1}$$

*where,*

$\rho$ is density of the liquid

g is acceleration due to gravity *assumed constant over this depth*

$P_{(due\ to\ liquid\ depth)}$ is pressue at depth 'h' due to weight of liquid

**Mass** is the quantity of matter. For a finite amount of fluid,it is independent of temperature & pressure of the fluid. **Weight** is the product of mass and acceleration due to gravity at that place. Numerically mass & weight are interchangeable, like for example, 10Kg mass is equivalent to a weight of 10Kgf.

**Volume** is the space holding the fluid. If a cylindrical tank of dia 2meters and a height of 1meter is completely filled with water; the volume of water is $\pi m^3$. Effect of pressure on volume of liquids is negligible,

***Becareful while choosing reference temperature***

[5]Though there is no true incompressibility, small compressibility can be neglected for liquids

| **Temperature** | **Absolute pressure** | **Establishing entity** |
|---|---|---|
| **$^oC$** | **kPa** | |
| 0 | 100.000 | IUPAC |
| 0 | 101.325 | NIST, ISO 10780 |
| 15 | 101.325 | ISO 13443 |
| 20 | 101.325 | EPA, NIST,NTP |
| 25 | 100.000 | SATP |
| 25 | 101.325 | EPA |
| **$^oF$** | **psi** | **Establishing entity** |
| 60 | 14.73 | EGIA,OPEC,US EIA |

Table 1: Standard Temperature & Pressure

NTP standard is used for calculations involving air, density @ NTP = $1.204 Kg/m^3$

but temperature has a significant effect on volume.The increase in volume of liquid per unit rise of temperature depends on its coefficient of expansion. Whenever measuring liquids at different locations, their volumes should be corrected to a standard temperature.

**The Jargon !** There are many different definitions of reference temperature and pressure world wide. Different organisations use different values at different context. Some popular values are here included.So while referring we have to use a common reference acceptable to the industry we serve. Also important is that we have to always refer to the same standards as we work with the project. In pipeline transportation a **'line fill volume'** of the pipeline refers to the volume contained between two points of isolation. This is equal to the internal cross sectional area of the pipeline times the distance between the point.
Volume of liquids(*or gases*) varies with temperature, so in pipeline tranportation over long-distances there could be discrepancies between the inlet flow and the outlet flow volumes even with no intermediary losses or injections.This is due to the change in temperature of the process fluid from inlet to outlet.

**Density** of liquids(*or gases*) is its mass per unit volume. This is referred to as mass density. The weight density is its weight per unit volume also called *specific weight.*As density is dependant on volume, with increasing temperature density decreases and *vice-versa*.

$$\rho_T = \rho_{15}(1 - \gamma(T - 15)) \tag{2}$$

*Where*

$\rho_T$ → density of liquid @ temperature $T^oC$

$\rho_{15}$ → density of liquid @ temperature $15^oC$

$\gamma$ → coefficient of density variation for temperature

**Specific Weight** It has a units of $(\frac{\mathbf{Newton}}{\mathbf{m^3}})$ and is dependent on the temperature of the fluid. Its significance in pipelines systems is that the product of specific weight and line fill volume gives the weight of the fluid in the section. It is important while calculating the loads on the piping system. Specifi Weight varies with the temperature of the fluid.

**Specific Gravity** of a liquid is the ration of its density to the density of fresh water at the same temperature. It is a dimensionless quantity and is used to express its heaviness relative to fresh water. **API Gravity** ($^oAPI$) is normally used for crude oils & petroleum products which is an ASTM laboratory scale which is the ratio of density of the oil to the density of water @ 60$^o$F. Density of water at this temperature is taken as 10.

$$SG_T = SG_{15} - \alpha(T - 15) \tag{3}$$

$$APIGravity = \frac{141.5}{SG_{60^oF}} - 131.5 \tag{4}$$

*Where*

$SG_T$ is specific gravity @ $T^oC$

$SG_{15}$ is specific gravity @ $15^oC$

$\alpha$ is coefficient for variation of specific gravity of the liquid

$SG_{60^oF}$ is specific gravity of liquid at $60^oF$ for $^oAPI$ scale

**What if i have mixtures ?** Specific gravity like Dalton's law of partial pressure exhibit the same analogy. *Specific gravity of a blended oil is sum of volume proportion of its consituent's specific gravity.*

$$SG_{mixture} = \nu_1 SG_1 + \nu_2 SG_2 + \cdots + \nu_n SG_n \tag{5}$$

*where*
$SG_{mixture}$ is the specific gravity of the blended oil consisting of 'n' individual types of oils.

$$\nu_i = \frac{v_i}{\sum_{i\to 1}^{n} v_i} \tag{6}$$

$v_i$ is volume of $i^{th}$ constituent $\nu_i$ is volume fraction of $i^{th}$ constituent

**Transmission of Fluid pressure** If some pressure is applied to the surface of a fluid, it gets transmitted to all points of the fluid. In a closed container if pressure is increased on a homogeneous liquid surface, pressure at all points inside the liquid experience the increase in pressure by the same magnitude.

**Absolute Pressure** at any point in a homogeneous fluid is equal to its specific weight times the depth of the point from surface added to the pressure acting above the surface of the fluid.

**Gauge Pressure** is the pressure displayed by a normal pressure gauge *which is zero set at standard atmospheric condition.* Gauge Pressure reflects the pressure exerted by the fluid above atmospheric pressure. *Also,*a vacuum gauge reflects the pressure below atmospheric pressure *sometimes wrongly shown as negative pressure.* Mathematically, absolute pressure & gauge pressure are related as -

$$P_{abs} = P_g + P_o \tag{7}$$
$$P_{abs} = P_o - P_{vg} \tag{8}$$

*where,*

$P_{abs}$ is Pressure absolute

$P_g$ is Gauge pressure

$P_o$ is Atmoshperic Pressure

$P_{vg}$ is Vacuum gauge pressure

**Pressure** of a fluid is same at all points on a horizontal plane inside a homogeneous fluid (*Pascal's Law*). Earlier we had a formula 1 for pressure at a depth 'h' by liquid alone. The formula gives gauge pressure if the free surface of the liquid is open to ambient atmosphere. But if same liquid is enclosed in a container at a pressure '$P_o$' above its free surface then, the pressure at the depth 'h' will be

$$P_{(@depth\,'h')} = P_o + \rho g h \tag{9}$$

**Thrust** over a plane immersed in a fluid is the summation of product of pressure acting on each infinitesimal elements of the plane.
Mathematically,

$$T_f = \sum_{i \to 1}^{n} (p_i a_i) \tag{10}$$

*where*
$T_f$ is the thrust force on the plane

**Viscosity** of a fluid is the property which describes its characteristics to flow, in otherways the shearing of each plane of the fluid with respect to the other. Fluids which are more thicker and hesitant to flow are more viscous. It is depenadant on temperature of the fluid. More the temperature lesser is its viscosity for a liquid, & just opposite for gases!. Though it is dependent to a very lesser extent on pressure, for flow in pipelines this variation is often neglected. Variation of viscosity with temperature is non-linear and both are plotted using semi-logarithmic scales. The variation of kinematic viscosity with temperature is shown below equation[6] no. 11.

$$\log_{10}\log_{10}(\nu + 0.7) = A - Blog_{10}T \tag{11}$$

for kinematic viscosity in the range of 2 x $10^7$ to 2 cSt *where*

$\nu$ is Kinematic viscosity

T is the temperature of the liquid in Kelvins

A,B are some constants specific to the liquid

**Viscosity of blended products** cannot be determined by using the weighted proportionate of the constituents. In the case of blending liquids of different viscosity an explanation of *Wright method* of ASTM D341 can be used. Alternatively the below mentioned Blending Index method[7] can be used. The procedure involves calculating blending indices.

$$H = 40.073 - 46.414\log_{10}\log_{10}(\nu + B) \tag{12}$$
$$B = 0.931(1.72)^{\nu} \text{ for } 0.2 < \nu < 1.5 \tag{13}$$
$$B = 0.6 \text{ for } \nu \geq 1.5 \tag{14}$$
$$H_{blend} = H_1(\%v_1) + H_2(\%v_2) + \cdots + H_n(\%v_n) \tag{15}$$

*Where*

H, $H_1 \cdots H_n$ are blending indices of constituent liquids

$H_{blend}$ is blended mixture blending index

B = constant for the liquid

$\nu$ = kinematic viscosity cSt

$\%v_1, \cdots, \%v_n$ are percentage of constituents in blend.

[6]MacCoull Neil, lubrication, The Texas Company, New York, June 1921
[7]Liquid Pipeline Hydraulics by E Shashi Menon

First the *blending index* for each of the constituent liquids should be evaluated using equation no. 12. Then the *blending index* for the mixture is evaluated applying for proportionate mixing equation no. 15. Finally the kinematic viscosity for the blended mixture is calculated.

**Vapour Pressure** All liquids exist at some equilibrium with the atmosphere above its surface. If you consider a closed tank with partly filled liquid, the atmosphere inside the tank will be at equilibrium to the liquid. The pressure of the vapor exerted on the liquid surface is called the vapour pressure of the liquid at the specific temperature. If additional pressure is exerted onto the vapour some part of it will change state to liquid. On the other way round, a decrease in pressure exerted on the vapour will cause some liquid to flash off to vapour. Similar behaviour can be noted with Temperature as well. An increase in temperature causes its vapour pressure to increase and a reduction in temperature causes reduction of vapour pressure. At the boiling point of a liquid its vapour pressure equals that of atmospheric pressure. Vapour pressure is very importnat topic in pipeline systems, vapour implosion can damage pumps, piping & throttling devices. A standard vapour pressure at $100^oF$ is called its *Reid vapour pressure.*

# 5 Basics of Pipeline Flow

## 5.1 Conservation of Mass

If 'M' is the mass of the liquid which enters a pipeline then the total mass of the liquid which should come out from all the outlets should be 'M' We have some relations on the basis of this concept depending upon how we can represent *(manipulate)* the mass of a liquid.

$$\begin{aligned} Vol \textbf{ x } \rho &= constant \\ A \textbf{ x } \mathsf{V} &= constant \\ Vol \textbf{ x } SG &= constant \\ A \textbf{ x } \mathsf{V} \textbf{ x } \rho &= constant \end{aligned} \tag{16}$$

*where*

A is the cross section area of the pipeline

$\mathsf{V}$ is the average flow velocity of the liquid

$\rho$ is the density of the liquid

Vol is the volume of the liquid

If you carefully look at the left hand side quantities of the equation 16 are all carefully manipulated quantities using conservation of mass; applicable for same homogeneous liquid in a pipeline. The relation of cross-sectional area to speed is called *equation of continuity*.

## 5.2 Conservation of Energy

Famous bernouli's equation on fluid flow is based on the energy conservation. The loss of velocity head reflects as potential energy while considering fluid flow in a pipe section of varying heights and cross sectional area. *Or conversely potential energy gets transformed as velocity head.*

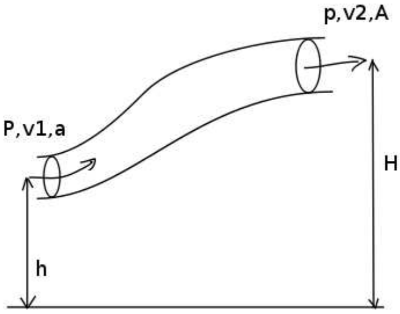



Fig. Krita Illustration for Bernouli's Equation

$$P + \frac{1}{2}\rho V_1^2 + \rho g h = p + \frac{1}{2}\rho V_2^2 + \rho g H \tag{17}$$

$$P + \frac{1}{2}\rho V^2 + \rho g h = constant \tag{18}$$

*where,*

**At Inlet:** Pressure = P, velocity = $V_1$, height above datum = h

**At Outlet:** Pressure = p, velocity = $V_2$, height above datum = H

$\rho$ is the density of the fluid, while g = acceleration due to gravity, *(assumed constant)*

*practically,*

$$\begin{aligned}
&Pressure_{(head\, of\, pump)} + \frac{1}{2}\rho V^2_{(delivery\, velocity)} = \\
&\quad Pressure_{(@reference\, point)} \\
&\quad + \frac{1}{2}\rho V^2_{(velocity\, @\, reference)} \\
&\quad + \rho g H_{(diff.\, in\, head\, between\, delivery\, point\, and\, reference\, point)} \\
&\quad + \sum (head\ losses)_{frictional,\, heat,\, turbulance,\, \cdots}
\end{aligned}$$

Equations 17 & 18 are the result of works by Daniel Bernouli, they are based on conservation of energy of the fluid entering the section and leaving the section. $\frac{1}{2}\rho v^2$ is the kinetic energy associated with the fluid and $\rho gh$ is the potential energy associated with the fluid. When the above relation is divided by the density of the fluid & 'g' acceleration due to gravity, the factors $\frac{Pressure}{\rho g}$ is called *'pressure head'*; $H$ is called *'gravity head'* & $\frac{V^2}{2g}$ is called *'velocity head'*

### 5.2.1 Validity of Bernouli's relation

Mach number is a dimensionless quantity which is

$$M = \frac{Speed\ of\ gas}{speed\ of\ sound\ in\ the\ gas} \tag{19}$$

$$speed\ of\ sound\ in\ gas = \sqrt{\left(\frac{C_p}{C_v}\frac{R}{M_m}T\right)} \tag{20}$$

*where,*

$C_p$ & $C_v$ are specific heats of the gas at constant pressure & volume respectively *for air* $\frac{C_p}{C_v} = 1.4$

R is universal gas constant *8.3144621* $\frac{Joule}{Mole.Kelvin}$

M$_m$ is molecular mass of the gas, *for air 28.98*

Bernouli's relation when applied to gases hold good only for mach numbers upto 0.3, above which issues of compressibility of the gas become significant.

## 6 How does a liquid behave during a flow ?

Fluid Flow is fluid in motion, our present area of discussion is flow in pipes & tubes. When we studied newton's laws of motion, we assumed they are applied to *'rigid particles'*. For rigid bodies, we assumed an equal mass of the body located at its center of gravity & applied the newton's laws of motion as applied to particles. But fluid flow is different, in the sense fluid is an elastic medium. We cannot intrepret the flow using particle or rigid body laws of motion equations. In the following pages, some calculations involving fluid flow in pipelines are presented. We introduce certain factors which affect flow and apply either conservation of mass or energy relations and complete the tasks. For a critical system, where cetain assumptions may lead to errors, finer approaches using computationalfluid dynamics with visualization produces more accurate results. Fluid flow can be broadly classed as *laminar flow* & *turbulent flow.*

**Laminar Flow** A flow condition in which all the particles in the fluid (*within a thinlayer*) move in a straight line, this is a desired flow condition of a pipeline flow.

**Turbulent Flow** A flow condition in which the particles in the fluid move irregularly, this is not a desired flow condition in pipelines.

**Transitional Flow** Between a laminar flow & turbulent flow, there exists an intermittent flow condition called '*transitional flow*'. The flow conditions here are uncertain and are a blend of each characteristic.

In a pipeline the fluid in contact with the wall of the pipe travels very slowly due to friction. The fluid velocity increases as we move towards the center of the pipe. Fluid travels in layers of increasing speed, reaching maximum velocity at the center of the pipe.

## 6.1 What are pipeline losses ?

Keeping a traditional picture in mind engineers over time have classified the pipelosses broadly as

1. Major losses - Frictional losses in pipelines.
2. Minor losses - (*pressure drop across valves, fittings, bends, filters, entrance & exit losses).*

Pressure loss in a pipeline depends on the flow, density, viscosity, velocity, surface roughness of the pipe, it is a manifestation of energy loss. The classification is based on assumption of fluid transportation over great lengths, where friction of a pipeline length is a major contributor to the losses. Pipeline fittings and components are fewer compared to the large length of piping. But in a ship or marine vessels, the classification is just a terminology as the lengths of piping is smaller and pipeline fittings and components are many in number. The pressure losses across such fittings or components is a significant loss.

## 6.2 Major Losses

Before we proceed to calculate how we can calculate major friction losses in a pipeline, we take a look at a dimensionless quantity 'Reynold's Number'.

### 6.2.1 Reynold's Number

In a pipeline consider a cross-section plane the velocity of the fluid 'V' is not same for the entire diameter. This is due to friction and viscosity of the liquid, let us call this velocity gradient as $\frac{dV}{dx}$. Then the rate of change of this velocity gradient is $\frac{d^2V}{dx^2}$ is the acceleration of seperation of the fluid

layers. **Reynold's Number ($R_e$)** is a dimension less number which gives a ratio of *inertial* forces to *viscous* forces. For a liquid of specific gravity '$\rho$' & dynamic viscosity '$\mu$' in a pipeline of diameter 'D',
Inertial forces for a pipe line cross-sectional plane is:

$$\rho \times V \times \frac{dV}{dx} \tag{21}$$

Viscous forces for the same sectional plane is:

$$\mu \times \frac{d^2V}{dx^2} \tag{22}$$

$$\begin{aligned} R_e &= \frac{\rho V \frac{dV}{dx}}{\mu \frac{d^2V}{dx^2}} \\ &= \frac{\rho V \frac{V}{D}}{\mu \frac{V}{D^2}} \\ &= \frac{\rho V D}{\mu} \\ &= \frac{VD}{\nu} \end{aligned} \tag{23}$$

*Where, $\nu$ is kinematic viscosity*

$$\nu = \frac{\mu}{\rho} \tag{24}$$

Reynold's number is important for checking flow in a pipeline.

1. The flow is laminar if $R_e < 2000$
2. The flow is critical if $2000 < R_e < 4000$
3. The flow is Turbulent for $R_e > 4000$

### 6.2.2 Head loss due to friction

Pressure losses due to friction in a pipeline is given by Darcy-Weisbach equation

$$h_f = f_D \times \frac{L}{D} \times \frac{V^2}{2g} \tag{25}$$

*where,*

$h_f$ is head loss due to friction

$f_D$ dimensionless friction factor known as Darcy-Weisbach friction factor. Consisting of $\pi$, Reynolds number & surface roughness of the pipeline internal surface.

L is length of the pipe & D is pipeline internal diameter

For laminar flow, Darcy-weisbach friction factor is related to Reynold's number as

$$f_D = \frac{64}{R_e} \tag{26}$$

The above relation is independent of the pipeline internal surface roughness for laminar flow. In case of turbulent flow, $f_D$ depends on Reynold's number and the pipe internal surface *relative roughness*. Relative roughness = $\frac{e}{D}$, where 'e' is called roughness height & 'D' is internal diameter of the pipeline. The darcy-weisbach friction factor can be calculated by the following methods:

1. **Moody's diagram** - *a graphical technique*

2. **colebrook - white equation** - This is a trial & error method of approximately calculating the friction factor. It requires us to assume a value and improvise upon the calculations to arrive at some accuracy for friction factor.

$$\frac{1}{\sqrt{f_D}} = -2\log_{10}\left(\frac{\frac{e}{D}}{3.7} + \frac{2.51}{R_e\sqrt{f_D}}\right) \tag{27}$$

3. **swamee-jain equation**

$$f_D = \frac{0.25}{\left[\log_{10}\left(\frac{\frac{e}{D}}{3.7} + \frac{5.74}{R_e^{0.9}}\right)\right]^2} \tag{28}$$

**Moody's Diagram** Friction factor from moody's diagram can be calculated in the following steps

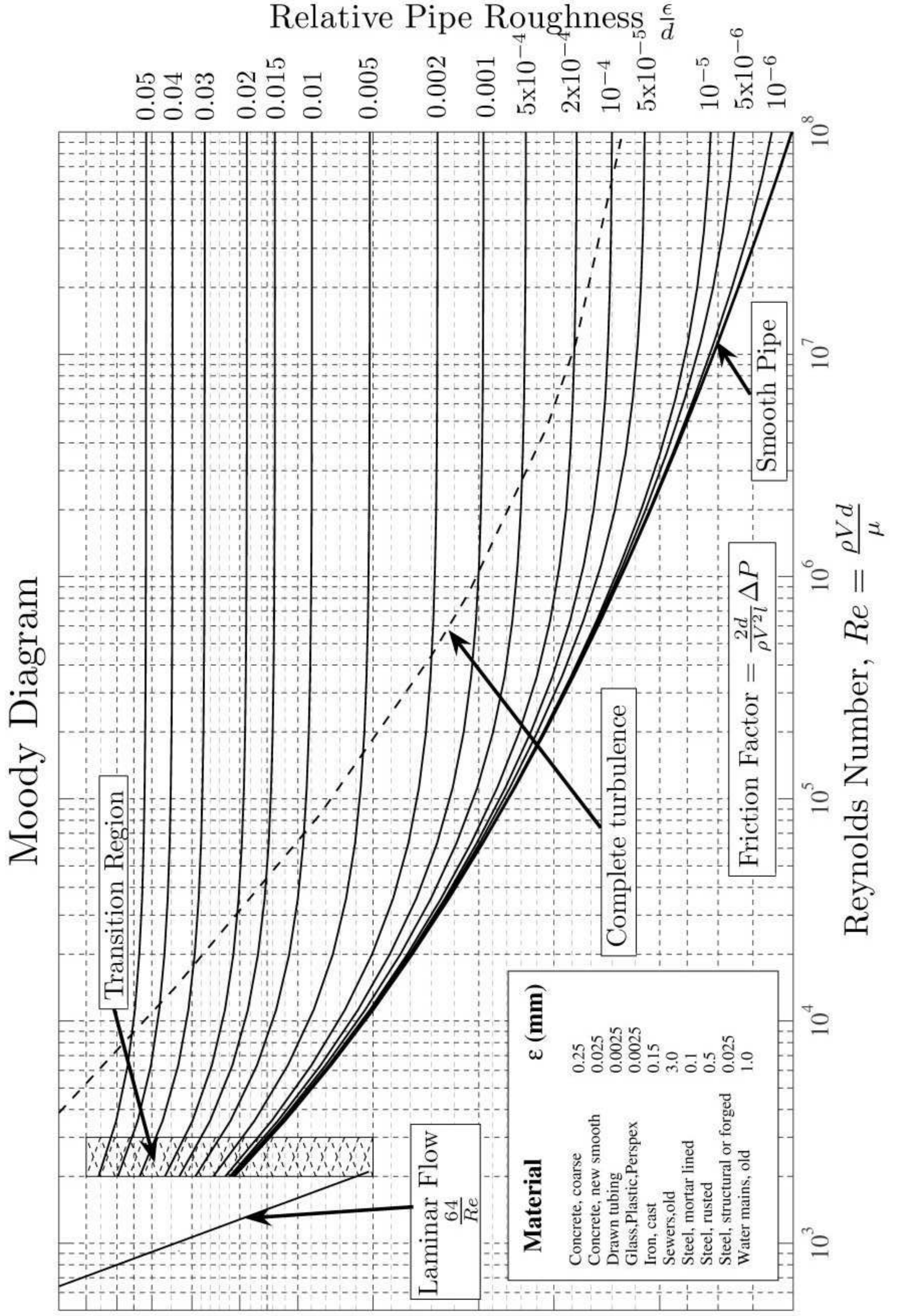
1. Find out roughness height 'e' for the specific material from tables.

2. calculate Relative pipe roughness $\frac{e}{D}$; draw a horizontal line through this value up to cut the first smooth pipe curve.

3. calculate Reynold's number, draw a vertical line through this value.
4. mark the point of intersection of the line through Reynold's number & the smooth pipe curve, draw a horizontal line from this intersection point to the friction factor.
5. The point where the above horizontal line cuts the friction factor line is the value of friction factor.

The equations discussed above are not valid if the flow is critical, as the flow is uncertain, in such a case calculations can be less accurately made using turbulent flow.

(source: wikipedia)



| **Material** | **Roughness(mm)** |
|---|---|
| Drawn Tubing, Glass, Plastic | 0.0015-0.01 |
| Drawn Brass, Copper, Stainless Steel (New) | >0.0015-0.01 |
| Flexible Rubber Tubing Smooth | 0.006-0.07 |
| Flexible Rubber Tubing Wire Reinforced | 0.3-4 |
| Stainless Steel | 0.03 |
| Wrought Iron (New) | 0.045 |
| Carbon Steel (New) | 0.02-0.05 |
| Carbon Steel (Slightly Corroded) | 0.05-0.15 |
| Carbon Steel (Moderately Corroded) | 0.15-1 |
| Carbon Steel (Badly Corroded) | 1-3 |
| Asphalted Cast Iron | 0.1-1 |
| Cast Iron (new) | 0.25 |
| Cast Iron (old, sandblasted) | 1 |
| Sheet Metal Ducts (with smooth joints) | 0.02-0.1 |
| Galvanized Iron | 0.025-0.15 |
| Wood Stave | 0.18-0.91 |
| Wood Stave, used | 0.25-1 |
| Smooth Cement | 0.5 |
| Concrete Very Smooth | 0.025-0.2 |
| Concrete Fine (Floated, Brushed) | 0.2-0.8 |
| Concrete Rough, Form Marks | 0.8-3 |
| Riveted Steel | 0.91-9.1 |
| Water Mains with Tuberculations | 1.2 |
| Brickwork, Mature Foul Sewers | 3 |

Table 2: absolute roughness of pipe materials (*courtesy www.nativedynamics.com.au*)

### 6.3 Minor Losses

When ever a fitting or a pipeline component is encountered, there is a frictional loss across it. Different fittings contribute to different losses. Pressure loss occurs when ever liquid passes through a bend, a reducer, gate valve, globe valve, a check valve or a butterfly valve. Though the drop in pressure is different for different valves, they can all be represented in a common relation *as loss of velocity head.*

**Pressure drop across valve or a fitting (*component*)**

$$h_m = k \times \frac{V^2}{2g} \tag{29}$$

*where*

k is Resistance coefficient (*fitting or component specific*)

V is velocity of the fluid ; & g is acceleration due to gravity

**Pressure drop across an expansion piece of a pipeline** Fittings like pipeline reducers are often used to join pipelines of different cross sectional area. When a flow encounters such a fitting, the pressure loss can be represented as: for $V_1$ velocity before change of cross section & $V_2$ velocity after change of cross sectional area,

$$h_m = k \times \frac{(V_1 - V_2)^2}{2g} \tag{30}$$

*for gradual expansion of the cross sectional area*

$$= \frac{(V_1 - V_2)^2}{2g} \tag{31}$$

*for sudden expansion of cross sectional area*

**Pressure drop across a sudden contraction of pipeline cross section area** When a fluid first breaks into the smaller section from the bigger cross section, a contraction of the fluid flow is formed. This is smaller than the smaller cross section of the component and shortly after in the downstream widens to fill the smaller crossection of the component. This contraction of area together with bigger cross section diameter & small cross section diameter determines the resistance coefficient. The head loss in this case is given by:

$$h_m = k \times \frac{V_2^2}{2g} \tag{32}$$

*where $V_2$ is fluid velocity in the reduced cross section of the fitting* &

$$k = \left[\frac{1}{C_c} - 1\right]^2 \tag{33}$$

$C_c$ is $\frac{A_c}{A_2}$ but also $C_c$ is dependent on ratio $\frac{A_2}{A_1}$

| $\frac{A_2}{A_1}$ | 0 | 0.04 | 0.16 | 0.36 | 0.64 | 1.0 |
|---|---|---|---|---|---|---|
| k | 0.5 | 0.45 | 0.38 | 0.28 | 0.14 | 0 |

$A_2$ is cross section area of reduced section, $A_1$ is cross section area of pipeline before contraction

$A_c$ is cross sectional area of the contraction, which fluid undergoes

**Entrance & Exit losses** Similarly entrance & exit losses can be evaluated by the following equations

$$h_{entrance} = k_{enter} \times \frac{V^2}{2g} \tag{34}$$

$$h_{exit} = k_{exit} \times \frac{V^2}{2g} \tag{35}$$

$K_{enter}$ varies from 0.8 to 0.04 depending on the entrance geometry

| **Geometry** | **k** |
|---|---|
| protruding pipe | 0.8 |
| flush end sharp edge | 0.5 |
| flush slightly rounded | 0.2 |
| flush well rounded | 0.04 |

$K_{exit}$ is 1

**Minor losses with valves, bends & fittings** The minor losses associated with valves & fittings are evaluated by the product of resistance coefficient ($k$) with the velocity head. Resistance coefficient for different types of fittings and valves are made available by the manufacturers.

$$h_m = k \times \frac{V^2}{2g} \tag{36}$$

# 7 Starting with pumps

Pumps impart mechanical energy to fluids, they cause flow or circulation in a pipeline. Pumps are classified as either 'Positive displacement' or 'Rotodynamic' depending on their design & operation. They essentially consists of fixed casing & a dynamic working component. For a rotodynamic pump the dynamic component is the impeller which rotates at a rpm, for a positive displacement pump it is either rotating profiled rotor,*like a screw, gear or a helix* or a sliding piston.

## 7.1 Positive Displacement

These are the pumps which are designed to deliver a fixed quantity ( *Volume*) each cycle. The pumps displace a fixed volume from suction to delivery side each cycle or stroke. They have a variable deliver head and a constant flow rate & should never be operated with closed or restricted sicharge piping. They are normally delivered with a built in relief valve across inlet and outlet for the pump ports. Some of the common types of positive displacement pumps which have a rotating elements are screw pumps, gear pumps & trochoid pumps. Reciprocating single acting or double acting piston pumps are also commonly used positive displacement pumps.

## 7.2 Rotodynamic Pumps

Two popular types of pumps which fit into this category are '*Centrifugal pumps*' & '*Liquid ring pumps*'. Fluid is directed to the suction of the rotating impeller, fitted with radially profiled vanes. The fluid is accelerated radially in the impeller and is forced into involute shaped casing, which transfers some of the kinetic energy into pressure head. These pumps operate with a constant head their flow depends on the delivery head in the pipeline. These pumps can for short time, be run with a shut delivery. However running for more time will heat up the pump parts and the fluid and the pump may seize.

### 7.2.1 Liquid ring pumps

Liquid ring pumps are rotodynamic pumps in which an impeller churns a liquid with a constant suction head in an elliptical casing. The resulting pulses of the liquid in the elliptical casing does the pumping action. They are used in air ejection systems for vacuum systems or for priming a centrifugal pump.

### 7.2.2 Centrifugal pumps

When a centrifugal pump is not of flooded suction design, the liquid which is accelerated radially by an impeller *(displaced liquid)* is reponsible for creation of vacuum in the suction. This vacuum in turn is reponsible for fluid to fill the suction. The liquid forms an essential component responsible for functioning of the pump.

Centrifugal pumps are not self priming, they need a priming device or a flooded suction design for operation at starting. Priming device is an equipment fitted in centrifugal pumps suction for ejection of air & vapour. By removing air & vapour pressure drops in the piping and liquid fills the place. Centrifugal pump will now function.

## 7.3 Pumps related terminology & what they mean

**Capacity *flow rate*** This is the volume delivered by the pump in a given time. Units are $\frac{m^3}{hour}$

**Suction pressure** The pressure at which a fluid enters a pump, measured close to the suction of the pump. Units are *bar (gauge)*

**Discharge pressure** The delivery pressure of the fluid as it leaves the pump, measured close to the delivery port of the pump. Units are *bar (gauge)*

**Differential pressure** This is the difference between the discharge & suction pressures. If the suction pressure corresponds to vacuum then, suction pressure is added to the discharge pressure. Units are *bar (gauge)*

**Flooded suction** This is the term used to describe positive suction pressure, where a fluid readily enters suction of a pump. This is essential to prevent cavitation.

**Static head** In a static fluid the pressure difference between any two points is directly proportional to the vertical distance between them. This pressure is due to the weight of the fluid column, this is called static head when stated in lengths of column of a known fluid. Units are *meter*

**Static suction head** This is the difference in height between the fluid level and the center line of the pump inlet, on the suction side of the pump. Units are *meter*

**Static discharge head** This is the difference in the height between the fluid level and the center line of the pump inlet, at the discharge side of the pump. Units are *meter*

**Total static head** This is the difference between the static discharge head and the static suction head, also called *static head of the system*. Units are *meter*

**Friction head** Pressure drop due to frictional losses in the suction and delivery sides of the pump. The frictional losses are expressed in loss of head, units *meter*

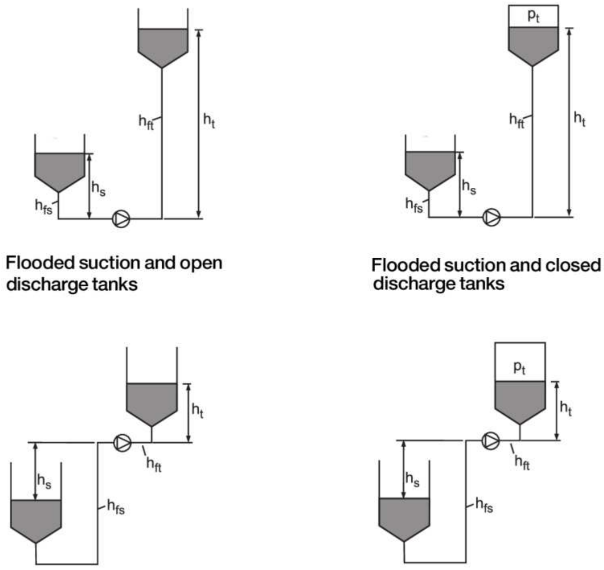
**Dynamic head** The energy required to set fluid in motion, the pressure lost in overcoming resistance to motion. Units are *meter*

**Total suction head** Static suction head minus dynamic head on the suction side of a pump is called total suction head. If the static suction head is negative, total suction head will be sum of static suction head & dynamic head. Units are *meter*

**Total discharge head** This is the sum of Static discharge head & dynamic head. Units are *meter*

**Total Head** It is the difference between the total discharge head & total suction head of the pump. Units are *meter*

**Mathematically the above definitions can be represented as:**




courtesy: Alfa Laval

$$\begin{aligned} \textbf{Total head H} &= H_t - (\pm H_s) &(37)\\ \textbf{Total discharge head H}_\mathbf{t} &= h_t + h_{ft} + p_t &(38)\\ \textbf{Total suction head H}_\mathbf{s} &= h_s - h_{fs} + (\pm p_s) &(39)\end{aligned}$$

$H$ = Total head

$H_s$ = Total suction head

$H_t$ = Total discharge head

$h_s$ = static suction head

$h_t$ = static discharge head

$h_{fs}$ = pressure drop in suction line

$h_{ft}$ = pressure drop in discharge line

$P_{ts}$ = pressure in the tank on suction

$P_t$ = pressure in the tank on discharge

## 7.4 Cavitation

Vapour pressure of a liquid is dependent on pressure & temperature, in other words; tendency of the liquid to vapourize increases with decreasing pressure or increasing temperature. In a process system, lowest pressure exists in the pump suction. So chances of vapour pockets developing in the liquid in pumps suction are high. And if the conditions favour, these vapour pockets generate and they are carried with the flow. These vapour pockets *'cavity'* in liquid implode on encountering higher pressures in the pump discharge or on impeller, casing & piping.

When this cavitation occurs it will result in the loss of pump efficiency. If the pumps are continued to operate with cavitation, mechanical damage to the pump & piping components will occur. Energy associated with such *'cavity'* implosions are high enough to destroy pump & piping system components whereever they occur. In design of any process or a piping system cavitation should be avoided.

### 7.5 Net positive suction head (***NPSH***)

The flow conditions of the fluid at the pump suction is vital to the operation of the pump. A well designed and installed pump should have the process fluid to enter the pump smoothly & at higher pressure to prevent '*cavitation*'. This pressure at which it is desirable for the fluid to enter the pump is called Net Positive Suction Head (**NPSH**). When looking at the data provided for a pump, NPSH required for satisfactory operation of the pump is quoted as '*NPSHr*' i.e Net Positive Suction Head *required* or Net Inlet Pressure Required *NIPR*. A designer will choose a higher pressure than this NPSHr for his / her design, which is called '*NPSHa*'. This is called Net positive suction head *available* or Net Inlet Pressure Available *NIPA*, this is done to guarantee that pump will not cavitate even for some tolerances in the piping design.

NPSHa or NIPA depends on:

1. Characteristic of the fluid, its viscosity
2. Inlet piping design, including any fittings or components provided *like filter, valve or any instrumentation*
3. Location of the suction tank, vessel or reservoir
4. Pressure above the fluid surface in the suction tank, vessel or reservoir

NPSHa or NIPA is what a pressure gauge reads close to the suction of the pump, when the pump is not in use. It is by design of the suction piping system, assumes a value.
Mathematically,

$$\text{NPSHa} = P_o \pm h_s - h_{fs} - P_{vp} \tag{40}$$

$P_o$ is pressure above the liquid level on the suction tank, vessel or reservoir

$h_s$ is the static suction head

$h_{fs}$ is the pressure drop due to friction in the suction piping

$P_{vp}$ is the vapour pressure at the temperature of the liquid

To prevent cavitation in pumps,

$$NPSHa > NPSHr \tag{41}$$

# 8 Pump sizing

## 8.1 What we need to know

### 8.1.1 Fluid characteristics

1. Fluid to be handled
2. Viscosity
3. Density
4. Fluid vapour pressure
5. Operating temperature
6. Purity or cleanliness of the fluid
7. Fluid behaviour
8. Toxicity of the fluid
9. Fluid compatibility with materials
10. Fluid reaction with atmospheric air

### 8.1.2 Performance data

1. Flow rate required
2. Discharge head
3. Suction conditions, like *flooded suction, NPSHa, ...*

### 8.1.3 Installation data

1. Location of installation, like *hull vibrations, ambient temperature, foundation, piping, submerged pumps*

2. Gland arrangement of the pump required (*mechanical, gland packing, seal cooling, flushing*)

3. Prime mover (*drive source available, diesel, hydraulic or electric*)

## 8.2 Power

Pumps impart mechanical energy to the fluid. The purpose is to either transport or cause circulation at a given flow rate, temperature & pressure. The energy imparted by the pump should generate required head, should compensate the pipeline losses. Rate of imparting this energy in a unit time is the power delivered by the pump or work done by the pump. This power is called *'Hydraulic Power'*.

### 8.2.1 Hydraulic power

The work done *'W'* on fluid whose density is '$\rho$', which produces a discharge head of *'H'* & a flow rate *'Q'* is

$$W = Q \times H \times \rho \times g \tag{42}$$

### 8.2.2 Power required by the pump

The input power available to the pump shaft should overcome the losses in the pump & should suffice the *Hydraulic power*. Hydraulic power is the useful power. The total power which should be available at the pump shaft is

$$W_{shaft} = \tau(Torque) \times \omega(\text{pump shaft rpm}) \tag{43}$$

## 8.3 Efficiency

Efficiency is defined for all pumps to judge how well an input energy is utilized in converting into head pressure (*pressure energy*). So it is dimensionless ratio. Some efficiencies are discussed only for rotodynamic pumps. The functioning of these pumps is different from the positive displacement pumps, so the efficiencies described are more appropriate for rotodynamic pumps.

### 8.3.1 Total Efficiency

The total efficiency is described for rotodynamic pumps. Total efficiency $(\eta_{total}) = \frac{Fluid\ Power}{Shaft\ Power} \times 100$
or,

$$\eta_{total} = \eta_{hydraulic} \times \eta_{mechanical} \times \eta_{volumetric} \tag{44}$$

### 8.3.2 Hydraulic Efficiency

In rotodynamic pumps such as a centrifugal pump, as the liquid enters the suction of the impeller, it encounters impeller and suffers change of direction. This presents some loss of power, which is called *'Shock loss'*. Then there are frictional and flow losses in the impeller & involute casing, these are called *'frictional loss'* & *'circulation loss'* correspondingly. Hydraulic efficiency reflects what fraction of the total head is lost as these losses.

$$\eta_{hydraulic} = \frac{head\ lost\ in\ pump}{total\ head} \times 100 \tag{45}$$

### 8.3.3 Mechanical Efficiency

Mechanical efficiency is a general efficiency of utilization of the pump shaft power input. This efficiency is applicable to all pumps, which is the percentage of utilization of input shaft power to mechanical energy of the fluid **in the pump**. The efficiency reflects losses in transmission, frictional at bearings, windage.

$$\eta_{mechanical} = \left(1 - \frac{mechanical\ losses\ in\ pump}{input\ power}\right) \times 100 \tag{46}$$

### 8.3.4 Volumetric Efficiency

In rotodynamic pumps there exists a difference in pressure between discharge & suction sides. Also all rotating mechanical components require clearances for smooth running. This presents certain loss of discharged fluid across impeller to suction, which is called *'leakage'*. Some power is lost due to these leakages and the efficiency which describes the energy lost in leakages is volumetric efficiency.

For Centrifugal pumps of a capacity 'Q' with a leakage of 'q' it is,

$$\eta_{volumetric} = \frac{Q}{Q+q} \times 100 \tag{47}$$

Positive displacement pumps work by shifting a fixed volume from suction to delivery sides each cycle. So the capacity of the pump would be $\frac{Volume\ per\ cycle}{cycles\ per\ hour}$. But in reality the discharge rate is less than this capacity due fact that in a given time fluid in suction may not fill the entire pump stroke volume. This factor is expressed as volumetric efficiency for positive displacement pumps.
In positive displacement pump of capacity 'q' which discharges at a flow rate 'Q',

$$\eta_{volumetric} = \frac{Q}{q} \times 100 \tag{48}$$

For positive displacement pump, the actual liquid filling its volume in each stroke is a function of the pump rpm. As more rpm means less liquid entering the pump as time for liquid filling will be short. Therefore following relation exists between pump rpm 'n' and its volumetric efficiency.

$$n = \frac{Q}{q \times \eta_{volumetric} \times 60} \times 100 \tag{49}$$

#### 8.3.5 Pump Efficiency

The pump efficiency is common to positive displacement pumps & rotodynamic pumps. This efficiency reflects how much pump input shaft power is converted to hydraulic power.

$$\eta_{pump} = \frac{Hydraulic\ power}{Input\ power} \tag{50}$$

or,

$$\eta_{pump} = \frac{Q \times \rho \times H \times g}{\tau \times \omega} \tag{51}$$

#### 8.3.6 Overall Efficiency

Overall efficieny of pumps is (Pump efficiency) × (Prime mover efficiency). Or,

$$\eta_{over\,all} = \frac{Hydraulicpower}{Input\ Power\ to\ prime\ mover} \tag{52}$$